SIMULTANEOUS CHILLED AND HOT WATER GENERATION USING SMELTER OFF-GAS WASTE HEAT

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Phase 0 Project Final Report

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OVERVIEW

This Phase 0 effort was undertaken to investigate the feasibility of utilizing waste heat from Aluminum smelters for the purpose of generating chilled water and hot water to be used for space conditioning and/or process heating applications. Based on discussions with Drs. Srinivas Garimella and Donald Ziegler of ALCOA, it was established that 800 Nm^3/sec of a hot gas stream is available for heat recovery. The gas is in fact a mixture of process gas at 960°C diluted with ambient air at a nine-to-one ratio. At this dilution rate, the mixture is available at a nominal temperature of 120°C. At the time of writing of the proposal, and certainly, during these subsequent discussions with ALCOA, it became clear that this waste heat stream is a very low grade heat source, bordering on the limits of feasibility for utilization in an absorption cycle. Although it is noted by ALCOA that the waste stream may have a heat content of up to 80 MW, this would require cooling the 120°C stream to such a low temperature, ~37°C, that it would not serve as a heat source along much of this temperature profile from 120°C to 37°C.

The objective of the present study, therefore, is to evaluate whether sources of such low grade heat can in fact be recovered in a beneficial manner. It is hoped that in view of the substantial projected increase in aluminum production over the next 15 years, coupled with the steep increase in energy costs currently being faced by industry, previously unviable techniques could be used now to advantage. In addition, the current concerns about global climate change and the environment are constantly leading to increasingly stringent requirements about emissions as well as source energy utilization efficiencies. This background serves as a favorable setting for the investigation of the feasibility of heat recovery through an absorption system.

SPECIFIC PERFORMANCE OBJECTIVES AND OPERATIONAL ENVELOPE

The discussion with ALCOA also revealed that these aluminum smelters are of varying capacities and operate in ambient temperatures ranging from -25°F to 95°F. This wide range of ambient conditions and waste heat capacities leads to the selection of ammonia-water as the working fluid instead of Lithium Bromide-Water, which is more commonly used in large tonnage commercial chillers. Furthermore, the low grade heat source dictates the use of a simple single-effect cycle – multiple-effect cycles yield higher coefficients of performance (COPs); however, they require higher temperature heat sources. There is also a "half-effect" cycle for specialized low grade applications, but in the present case, the additional complexity of that cycle might preclude its use. Based on these considerations, a baseline single-effect ammonia-water cycle is first developed here to accomplish the following:

- Generate a chilled water stream at approximately 45°F, to be used for building and office space cooling
- Use the reject heat from the condenser and absorber to generate a hot water stream that could be employed for building water heating needs, either in standalone mode, or to supplement dedicated water heaters

It is expected that if successful, the eventual implementation of such a system will increase overall process efficiencies, reduce the need for electrically driven air-conditioning and chilling equipment, reduce plant operation costs, and assist the aluminum smelter facility to achieve compliance with evolving energy efficiency and emissions regulations.

CYCLE DESCRIPTION

A representative schematic of the basic ammonia-water single-effect cycle is shown in Figure 1. (It should be noted that this schematic represents one basic set of connections between the various components. Alternate connections between components, such as varying absorber and condenser coolant flows from series to parallel and vice versa, can also be considered, depending on the details of the application, without changing the basic configuration of the ammonia-water cycle. Referring to Figure 1, the absorber saturated outlet condition is represented by state (1), with the subcooled outlet being state (2). This concentrated solution flows through the solution pump and exits at state (3). The solution is then recuperatively heated to state (4) at the desorber inlet. The corresponding saturated state at the desorber inlet is represented by state (21). Thermal energy is provided to the concentrated solution stream in the desorber using the waste heat gas stream entering at state (19) and leaving at state (20). The

dilute solution (7) and vapor (6) streams exit the desorber. The vapor stream flows through the rectifier, yielding concentrated ammonia vapor (8) and a reflux stream (9). The reflux stream combines with the dilute solution exiting the desorber and yields the solution at state (16), which enters the solution heat exchanger.

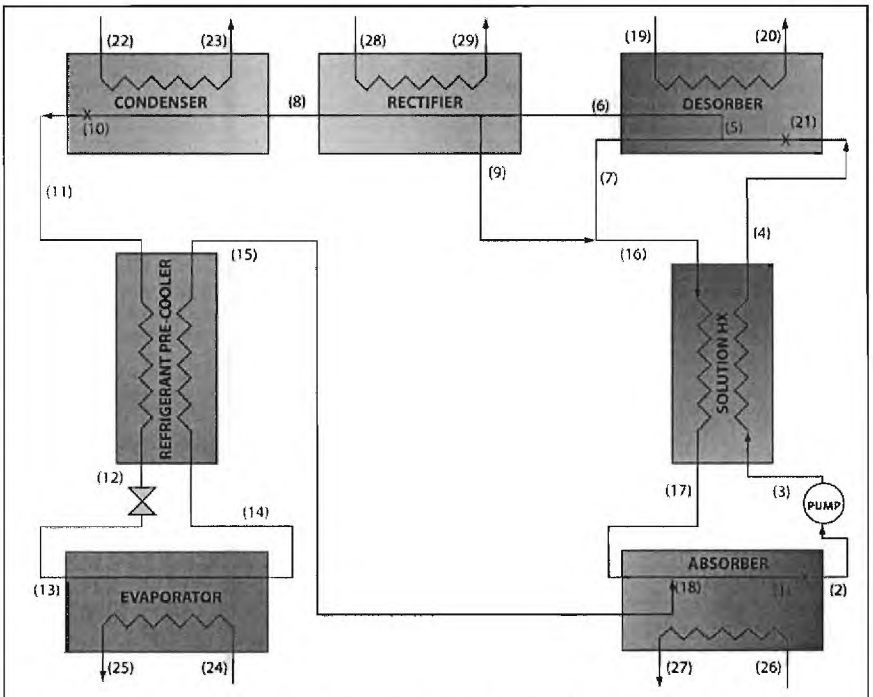

Figure 1. Absorption Cycle Schematic

Cooling for the rectifier is provided by a fluid entering at state (28) and exiting at state (29). This coolant could be air, glycol-water solution, or the concentrated solution exiting the absorber. The purified vapor is condensed in the condenser, with the saturated liquid state depicted by state (10) and the subcooled liquid outlet represented by state (11). Cooling for the condenser is provided by a fluid entering at state (22) and exiting at state (23). The ammonia exiting the condenser (11) is recuperatively cooled to state (12) in the refrigerant pre-cooler to reduce its enthalpy at the expansion valve, thereby increasing the cooling capacity. The ammonia then flows across the expansion valve, entering the evaporator as a two-phase mixture at state (13). The cooling load is provided to the evaporator by a stream entering at state (24) and leaving at state (25). The ammonia exits the evaporator as a two-phase mixture (14) of a higher quality and at a higher temperature than at the evaporator inlet due to the temperature glide induced by the residual water fraction in the ammonia. Upon exiting the evaporator, the ammonia flows through the refrigerant pre-cooler, receiving the heat rejected by the condenser outlet stream, and leaves the pre-cooler at state (15). This ammonia stream flows to the absorber, where it combined with the returning dilute solution (17) from the solution heat exchanger, yielding a two-phase mixture at state (18). As this mixture flows through the absorber, the vapor phase is absorbed

into the liquid phase due to heat rejection to the coolant entering at state (26) and exiting at state (27), thus completing the cycle.

Mass, species and energy conservation equations are solved for each component shown schematically in Figure 1. In addition to these thermodynamic balances, heat and mass transfer resistances in each component are addressed by specifying representative values of overall heat transfer conductance UA (desorber, condenser, absorber) or heat exchanger effectiveness ε (refrigerant pre-cooler and solution heat exchanger). It should be noted that these specifications are necessarily of a single-point type, and represent a first-order analysis of the overall system. However, they do capture, in adequate measure, the driving potentials required for the interactions between the streams exchanging heat and/or mass, and also the effects of irreversible heat exchange between the source/sink and the working fluid across the temperature differences that represent realistic component performances. Representing components with varying thermal capacities by a unique value of UA, ε or closest approach temperature difference CAT constitutes an approximation to facilitate a preliminary estimation of system performance.

BASELINE SYSTEM DESIGN CONDITIONS, CHOICES AND ASSUMPTIONS

On the basis of information received from ALCOA, the design conditions and a baseline system layout were selected. A schematic of this system is shown in Figure 2, along with key variables predicted by the computations for this baseline case. The system has the following features:

- Waste heat gas stream directly coupled to the desorber. This choice was made instead of first exchanging heat between the gas stream and a heat transfer fluid, which would then supply the heat to the ammonia-water mixture in the desorber. This is because such an intermediate heat transfer loop would introduce a temperature drop between the source and the absorption cycle, and is therefore not an optimal choice in instances such as the present case where the source temperature is already very low. Based on the above discussion, a waste gas stream inlet temperature of 248°F was chosen, at a flow rate of 7.5×10^6 lb_m/hr. It is also assumed that for this screening study, the properties of this fluid are well represented by those of air at similar conditions. This inlet temperature is in fact one of the critical parameters in this project. Even small changes in this temperature affect the system performance significantly. These effects are discussed after the presentation of the baseline results.
- A 50% ethylene glycol-water mixture is used as the coupling fluid for the evaporator. This fluid stream is chilled from a nominal inlet temperature of 55°F to an outlet temperature of 45°F, which is a temperature range representative of commonly used chillers. The hydronic fluid flow rate is chosen to be 1000 gpm. The implications of this choice and the effect of changing this flow rate are discussed in subsequent sections.

- The condenser and absorber are coupled to another hydronic fluid (50% ethylene glycol water mixture) stream in series. Thus, the hydronic fluid first receives the heat of condensation, and is then further heated in the absorber. Series coupling between the condenser and absorber is appropriate here because it will generate the hottest possible water stream compared to parallel coupling of these components. The ethylene glycol-water solution that serves as the hydronic fluid can also be changed to water, depending on the actual application under consideration. It was assumed that for the hot fluid generated in these two components to be of any practical use, the hydronic fluid should enter the condenser at 110°F and be heated by at least 20°F to a temperature of 130°F. (It is deemed that there are few heating applications for which a hot fluid temperature less than 110°F would be useful.) The flow rate of this hydronic fluid stream is assumed to be 1400 gpm. Again, this flow rate is dictated by the need to generate hot fluid for process utilization and result in at least a 20°F rise in temperature. A larger flow rate would reduce the temperature rise; however, system cooling performance (chilled water generation) would improve at higher hot hydronic fluid flow rates. These effects are discussed in a subsequent section.
- The rectifier uses the concentrated solution exiting the solution pump as the coolant. This makes rectifier cooling an internal ammonia-water process, rather than requiring yet another cooling fluid loop. Since cooling using concentrated ammonia-water solution is a recuperative process, it will also result in a marginally higher COP than a hydronic fluid cooled rectifier.
- The concentrated ammonia-water pumped solution flow rate is set at 1.05×10^5 lb_m/hr.

Initial Specification of Heat exchangers

For a preliminary analysis such as this, to enable the initiation of the computations, an approximate specification of the heat exchangers to be used for each component is necessary. Therefore, the heat exchangers are initially specified in terms of the closest approach temperature difference (CAT) that they can achieve between the two fluids. The larger the CAT, the smaller the size of the heat exchanger required, and vice versa. A large CAT, however, "eats into" the available temperature difference between source and sink in the cycle, and leads to lower system performance. In the present situation, due to the really low source temperature availability, the lowest possible CAT should be used. Of course, this implies larger heat exchanger sizes and capital cost. In the present analysis, the lowest reasonable CAT was used for each major component. The rationale here is that the potential chilling and water heating performance with the best possible heat exchangers should be evaluated first. If this performance is not deemed satisfactory, there would be little reason to choose smaller heat exchangers, because they would result in poorer system performance. Therefore, the CAT for the desorber, condenser, evaporator and absorber was chosen to be 5°F. This implies that each of these heat exchangers is able to bring the hot fluid outlet down to within 5°F of the cold fluid inlet temperature, or

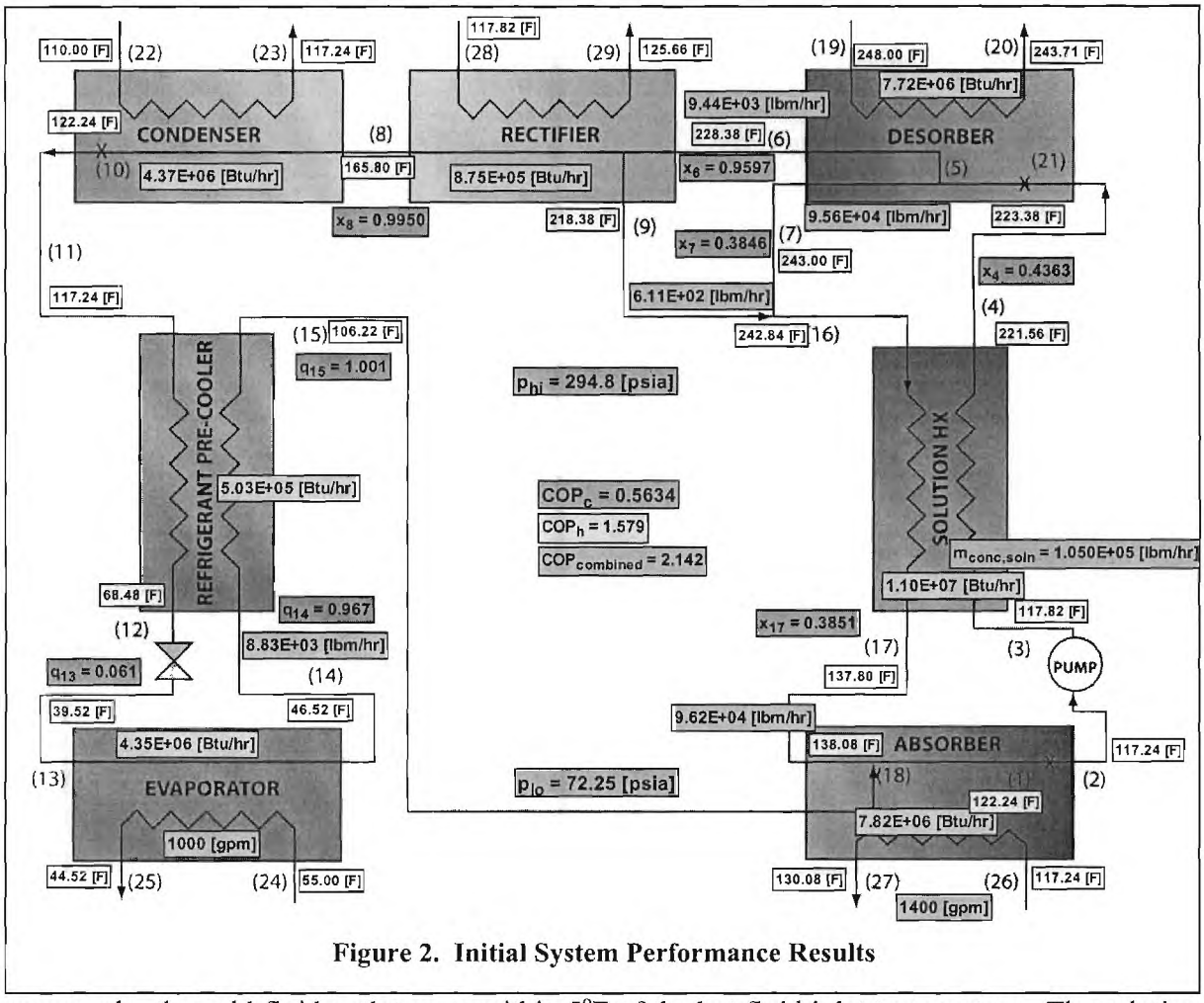

Figure 2. Initial System Performance Results

conversely, the cold fluid outlet up to within 5°F of the hot fluid inlet temperature. The solution heat exchanger and the recuperative refrigerant heat exchanger were assumed to have heat exchanger effectivenesses of 90%. These counterflow heat exchangers should be able to achieve such effectivenesses. After the system level computations and results are presented, approximate surface areas required to achieve these CATs and heat exchanger effectivenesses are presented.

INITIAL SYSTEM COMPUTATION AND PERFORMANCE RESULTS

Desorber

The waste gas stream enters the desorber at 248°F and is cooled to 243.71°F as it heats the ammonia-water solution entering the desorber. The concentrated solution enters the desorber at 43.63% ammonia at a temperature of 221.56°F. (Given a specified pumped concentrated solution flow rate, the concentration of the concentrated solution is determined by the absorber heat transfer specification.) It is first heated to the saturation temperature of 223.38°F, at which

point desorption starts. Dilute solution exits the desorber at a concentration of 38.47% and a temperature of 243°F. This dilute solution outlet concentration is determined by the corresponding saturation temperature (243°F), which is in turn set by the waste heat source temperature and the desorber CAT specification. Upon counterflow desorption, 9.44×10^3 lb_m/hr of ammonia vapor at a concentration of 95.97% and a temperature of 228.38°F is generated. It should be noted that the vapor was assumed to leave the desorber at a temperature 5°F greater than the saturated inlet temperature of the desorber, which accounts for the inequilibrium between the two counterflow streams (dilute solution and vapor.) The desorber heat duty is 7.72×10^6 Btu/hr (2.26 MW).

Some comments on the heat recovery from the waste gas stream are appropriate here. It is seen above that only 2.26 MW are recovered from the waste gas stream as opposed to the nominal potential for recovering 80 MW if the gas is cooled all the way to 37°C. This is true of any low grade waste heat recovery system, where any recovery at a low enough temperature, even if above the nominal ambient temperature, would not have any practical use. In this case, the amount that can be recovered is established by how high the outlet temperature of the dilute solution exiting the desorber must be. In this analysis, the dilute solution exits at 38.47% ammonia, which represents a change in concentration from 43.63% (of the concentrated solution) of about 5%. This change directly establishes the ammonia vapor generated for chilling purposes. This is a fairly small concentration change (or desorption rate). If the concentration change in the desorber were smaller, the dilute solution outlet temperature would be commensurately lower, allowing the waste gas stream to be cooled a little further. But a smaller solution concentration change also conversely implies a smaller vapor production rate, which is the object of the desorption process. Clearly, a higher source temperature would help considerably; however, it is understood that this is established by aluminum smelter operating parameters.

Rectifier

The vapor generated in the desorber is purified in the rectifier. Essentially, cold concentrated solution interacts thermally across the rectifier wall with the impure vapor, and in the process condenses out some of the water fraction in the vapor, yielding a vapor stream at a higher ammonia concentration. The purified vapor leaves at a specified concentration of 99.5%. It should be noted that this outlet concentration is a specified input value, with the assumption

that the component will be designed to meet this outlet concentration requirement. (It should also be noted that in ammonia-water heat pumps, a concentration of 99.95% and greater is desired at the rectifier outlet, so that the temperature glide in the evaporator is not excessive.) With these assumptions, the yield of purified ammonia vapor is 8.83×10^3 lb_m/hr, with a reflux flow rate of 611 lb_m/hr. The solution that serves as the coolant for this rectification process flows from the solution pump, entering at 117.82°F, and leaving at 125.67°F, representing a rectification load of 8.75×10^5 Btu/hr.

Condenser

The rectified vapor enters the refrigerant condenser at 165.80°F, condensing at a pressure of 294.8 psia. This pressure is established by the selection of the hydronic fluid inlet temperature of 110°F. The condensation heat load raises this temperature to 117.24°F, which then implies that the condenser saturation temperature must be at least 5°F above this temperature. The resulting condensation temperature of 122.24°F therefore sets the high-side pressure of 294.8 psia. This high-side pressure also participates in setting the ammonia temperatures in the desorber that were discussed above – for a given solution concentration, the higher the saturation pressure, the higher is the desorber saturation temperature, which then reduces the driving temperature difference between the waste gas stream and the ammonia-water solution in the desorber, and reduces the amount of heat that can be recovered from the waste stream.

It should be noted that in setting the condenser saturation temperature, consideration was given to the fact that the residual water fraction (0.05%) in the ammonia stream condenses out first preferentially within a small length of the condenser, followed by the condensation of the ammonia, such that the saturation temperature for the bulk of the condenser is assumed to be the liquid phase temperature of 122.24°F, even though in reality, this process is non-isothermal. A subcooling of 5°F is assumed at the condenser outlet. The resulting total condenser heat rejection load is 4.37×10^6 Btu/hr.

Recuperative Refrigerant Pre-Cooler

The condensed refrigerant enters a recuperative pre-cooler at 117.24°F, where it is cooled to 68.48°F by the refrigerant stream exiting the evaporator. A heat exchanger effectiveness of 90% is assumed for this component. The corresponding vapor stream is heated from 46.52°F to 106.22°F, for a heat duty of 5.03×10^5 Btu/hr. The advantage of placing this heat exchanger in

the system is that it enables the refrigerant to enter the evaporator at a lower enthalpy than the condenser outlet enthalpy, so that as the refrigerant expands across the expansion valve, very little flashing occurs, and the refrigerant is almost entirely in the liquid phase at the evaporator inlet. Thus, more of the latent heat of evaporation can be used for the chilled water.

Expansion Valve

The isenthalpic expansion valve reduces the refrigerant pressure from the high-side value of 294.8 psia to the low-side pressure of 72.25 psia, leading to flashing of the refrigerant to a quality of 6.1%. This two-phase refrigerant mixture enters the evaporator at a temperature of 39.52°F. It should be noted that the low-side pressure is established by the evaporator hydronic fluid temperature and its heat exchange capabilities (represented here by $CAT_{evap} = 5°F$).

Evaporator

The evaporator cools the 1000 gpm of hydronic fluid from 55°F to 44.52°F, for a heat duty of 4.35×10^6 Btu/hr (1.275 MW). An evaporator refrigerant temperature glide of 7°F is assumed. Unlike a pure refrigerant, an ammonia-water mixture evaporates at a temperature that increases as the vapor quality increases. Much of this temperature rise occurs at the end of the evaporation process. If the evaporator were required to produce refrigerant at 100% vapor quality, the refrigerant temperature would rise to 104.6°F. This would of course result in no cooling, because the hydronic fluid temperature would have to be even higher. By limiting the glide to 7°F, the refrigerant exits the evaporator at 46.52°F and a quality not of 100% vapor, but at a quality of 96.7%, i.e., almost completely vapor. If the glide were limited to much lower values, the refrigerant outlet quality would be very low, thus not making good use of the refrigerant generated in the desorber. The actual control of the evaporator temperature glide would be accomplished by a thermostatic expansion valve. Parametric analyses on the rectifier outlet concentration, the evaporator temperature glide, and the evaporator UA would be required to refine the optimal values for these parameters for the specific system under consideration.

Dilute Solution and Reflux Mixer

The reflux from the rectifier (611 lb_m/hr) is mixed with the dilute solution exiting the desorber (9.56×10^4 lb_m/hr) to yield a solution stream of 9.62×10^4 lb_m/hr at a concentration of 38.51%. It should be noted that the reflux is assumed to exit the rectifier at an inequlibrium of

10°F with the vapor stream entering the rectifier, which accounts for the mass transfer resistance in the rectifier.

Solution Heat Exchanger

The solution heat exchanger cools the dilute solution stream from 242.84°F to 137.80°F. In the process, it heats the concentrated solution exiting the rectifier from 125.67°F to 221.56°F. This solution stream enters the desorber to complete the solution circuit. A solution heat exchanger effectiveness of 90% is assumed for this component, yielding a heat duty of 1.10×10^7 Btu/hr.

Absorber

The dilute solution and refrigerant vapor streams are combined in the absorber. The solution-refrigerant mixture achieves a temperature of 138.08°F at the absorber inlet. The heat of absorption is then removed from the mixture to result in a concentration solution with x = 43.63% at a saturation temperature of 122.24°F. A subcooling of 5°F is also assumed for the absorber, resulting in a concentrated solution outlet temperature of 117.24°F. The absorber heat load of 7.82×10^6 Btu/hr is removed by the hydronic fluid stream. The hydronic fluid exiting the condenser enters the absorber at 117.24°F, and is heated to an outlet temperature of 130.08°F. This outlet temperature, therefore, accomplishes the objective of heating water in this system from 110°F to 130°F.

Solution Pump

The solution pump raises the solution from the low-side to the high-side pressure. A pump efficiency of 70% is assumed, resulting in a pumping power of 34.8 kW. The solution pump sets the solution flow rate of 1.05×10^5 lb_m/hr. It can be seen that the solution pump power consumption represents a miniscule portion of the overall system capacity, whereas in a corresponding vapor compression system, the compressor would be the major power consumption component. Additional parametric analyses on this flow rate would establish the optimal flow rate and the concentration split between the concentrated and dilute solution to achieve the most desirable performance for a specific system configuration. The solution exiting the pump flows to the rectifier.

System COP

The system described above yields a cooling mode COP (= Q_{evap}/Q_{source}) of 0.563, which is quite good for a low temperature heat source application. However, it must also be recognized that aggressive assumptions about heat exchanger capabilities (CAT = 5°F) were made to arrive at this system performance. If the heating performance is viewed as the desired objective, the heating COP (=$(Q_{condenser}+Q_{absorber})/Q_{source}$) is 1.579. However, since the stated desired objective is simultaneous chilling and heating of water, the more appropriate definition of COP here would be in terms of the total useful output divided by the source heat input. Thus, $COP_{effective}$ can be written as follows:

$$COP_{effective} = \frac{Q_{cooling} + Q_{heating}}{Q_{source}} = \frac{Q_{evaporator} + Q_{condenser} + Q_{absorber}}{Q_{source}} \quad (1)$$

$$COP_{effective} = \frac{1275\,kW + 3573\,kW}{2263\,kW} = \frac{1275\,kW + 1281\,kW + 2293\,kW}{2263\,kW} = 2.142 \quad (2)$$

Thus, for every MW of waste heat used, 2.142 MW of useful product (cooling or heating) is generated. This value is of course strongly dependent on the system configuration and component sizes assumed. It should be noted that this COP value does not include consideration of the energy required to run the hydronic fluid pumps or the small amount of energy expended in the solution pump. Inclusion of these quantities, heat losses and other realistic phenomena will result in somewhat lower COPs than this prediction.

OPERATING COST SAVINGS ESTIMATES

The cooling capacity discussed above can be used to estimate operating cost savings.

Savings from Chilled Water

For an equivalent vapor compression system, assuming a nominal COP of 3, to generate 1 MW of cooling, 1/3 MW of electrical input would be required. The cooling capacity achieved in this system would therefore require 0.425 MW of electrical input. At a nominal rate of 8 cents/kWh, every hour of operation would save $34.

Savings from Hot Water

Similarly, the heating capacity from this system can be translated into operating cost savings. Assuming a conventional gas to heat efficiency of 0.90, 135.5 therms/hr of natural gas

would be required to supply the heating generated by this system. At a representative cost of gas of $1.2/therm, this translates into a savings of $163/hr.

Pumping Power Consumption and Net Savings

An approximate estimate of the total power consumed in the solution pump and the two hydronic pumps can also be made. The pumping power would depend significantly on the local layout of the facility to which heating and cooling must be supplied. If it were assumed that the cooling fluid is pumped across a 25 psi pressure drop, while the heating fluid is pumped across a 40 psia pressure drop, and if it is further conservatively estimated that pump efficiencies for these pumps are only 35%, then the pumping power for circulating the chilled and heated fluids would be 31 and 70 kW, respectively. Summing the pumping power for the ammonia solution and these two hydronic fluid pumps, the total power consumption is 136 kW. The cost of providing this power, again at $0.08/kWh, would be $11/hr. In sum, the savings per hour of operation of this system, accounting for the power incurred in pumping, would be $197-$11 = $186.

Extrapolation to Annual Basis, and Plant/Location Based Uncertainties

Assuming that due to the highly thermally intensive processes in an aluminum smelter, the operation is conducted round the clock so that no thermal cycling losses are incurred, the savings per day can also be calculated in a straightforward manner. Thus, the total savings due to cooling and heating would amount to $4458 per day. If further extrapolated to an annual basis with allowance of a capacity factor of 75% for scheduled maintenance and downtime, the annual savings would be $1.2 million.

The above annual savings estimate will depend to a large extent on whether there is a local need on the plant premises for chilled water and hot water. This is a big unknown that depends very much on the balance of plant at each smelter site. If there is no viable end use for either chilled water of hot water at the plant site, these heating or cooling savings would not apply, especially because hot and cold fluid streams are not easily transportable over long distances without incurring significant heat loss and parasitic pumping power requirements. These are site specific considerations that are best addressed by ALCOA staff on a plant-by-plant basis.

Some major assumptions about rate structures were made in arriving at these cost savings estimates. Rates structures are a highly local phenomenon, and also involve complex short-term

and long-term agreements between the utility and the user, especially when the user is a large industrial customer such as an aluminum smelter. In addition, often electric rates are based on incremental capacity utilization and are subject to very large demand charges in addition to the /kWh charges. These would skew the operating cost predictions considerably, but generally toward higher cost savings. Therefore, it is best if the heating and cooling capacities are used by ALCOA staff in conjunction with their internal price structure information to estimate costs savings with better accuracy.

HEAT EXCHANGER SIZE ESTIMATES

Although the primary focus of the present study is an overall estimation of the cooling and heating capacities that can be provided by this waste heat recovery system, a very preliminary estimation of the surface areas required to implement this system was developed. As stated previously, reasonable CAT and heat exchanger effectiveness estimates were made to predict system state points. Once the system state points were computed as shown in Figure 2, these temperatures were used to compute component log-mean-temperature differences (LMTDs), which then yielded the UA required for each component to operate at this LMTD and transfer the heat transfer rate Q computed in the system analysis. Thus,

$$Q = UA \times LMTD \tag{3}$$

Additionally, based on the heat transfer process occurring on each side of each component (single-phase gas, single-phase liquid, phase-change heat transfer), approximate overall heat transfer coefficients, U, were selected. This (very preliminary) process then yields estimates of the effective surface areas required for each component:

$$A_{required} = \frac{UA_{required}}{U_{assumed}} \tag{4}$$

These effective surface areas (which could be provided using prime and finned surface areas in an actual heat exchanger) are shown below.

Component	Process	U (Btu/hr-ft^2-F)	UA (Btu/hr-F)	A (ft^2)
Desorber	Single-phase gas to phase-change	15	707,000	47,100
Evaporator	Single-phase liquid to phase-change	500	661,000	1,320
Condenser	Single-phase liquid to phase-	500	540,000	1,080

	change			
Absorber	Single-phase liquid to phase-change	500	1.23×10^6	2,450
Solution Heat Exchanger	Single-phase liquid to single-phase liquid	300	679,000	2,260
Refrigerant Pre-Cooler	Single-phase liquid to phase-change near dryout	250	31,700	127
Rectifier	Single-phase liquid to gas with liquid film	150	12,200	81

As can be seen from a quick review of these surface areas, the desorber (i.e., the actual waste heat recovery device) is by far the largest component in the system. This is because this heat exchange is between a gas and a phase-change process across a small temperature difference. One obvious reason for this large size is that the low LMTD for this heat exchanger arises from the very low temperature (248°F) gas stream being used as the heat source. In natural gas-fired heat pumps, the flue gas stream is typically available at ~2000°F, and its temperature can decrease by more than 1500°F as heat is extracted from the stream without imposing such a pinch temperature on the desorption process. In such cases, the low gas-side heat transfer coefficient is not of much concern, because the high LMTD compensates well for the low U, resulting in fairly small desorber surface areas. Gases have inherently low heat transfer coefficients that dominate the thermal resistance in such heat exchangers. Almost all of this surface area must be provided on the gas-side. This can be accomplished using large finned heat exchangers with a high fin density. The waste heat gas stream would flow across these fins, transferring its heat to the ammonia-water solution flowing inside the finned tubes. Innovative designs would be necessary to accomplish this heat exchange on the air side and counterflow desorption of the vapor from the concentrated solution flowing inside the tubes. Such a design exercise is beyond the scope of this preliminary feasibility assessment. It should, however, be noted that at such small temperature differences in the desorber, even a small increase in the source temperature would have a significant positive effect on the system, either by increasing heat recovery considerably for every 1°F increase in heat source temperature, or decreasing heat exchanger size requirements considerably while keeping system performance the same. A parametric analysis of the effect of source temperature on system performance is provided in the following section.

Other components in the system require much smaller surface areas because of the single-phase liquid or phase change heat transfer processes occurring on both sides of these heat exchangers.

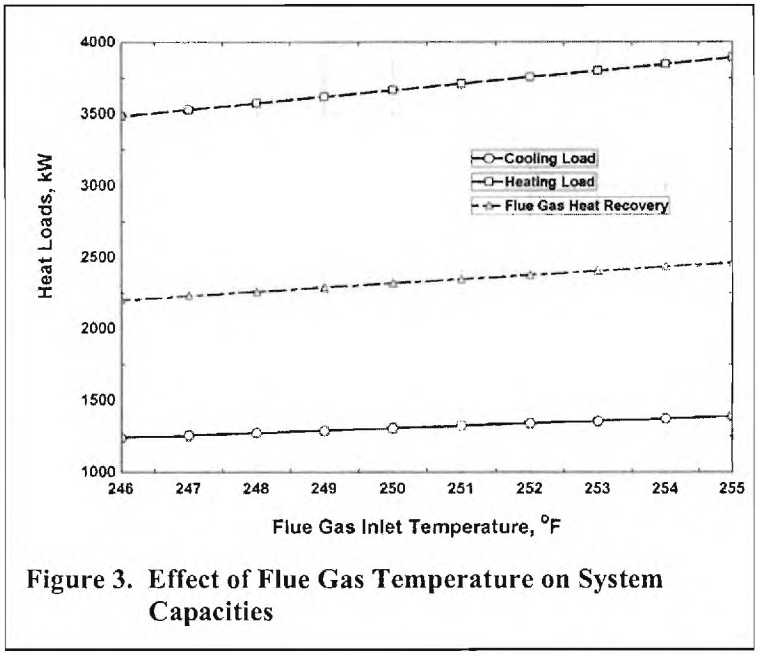

Figure 3. Effect of Flue Gas Temperature on System Capacities

PARAMETRIC ANALYSES

Effect of Flue Gas Temperature

As stated above, the heat source temperature is the most critical parameter in determining the performance of the heat recovery system. To quantify the influence of this parameter, several different analyses were conducted, and are summarized here. The analyses thus far have specified CAT values for each heat exchanger, and determined the surface areas necessary to obtain these CATs. In the following analyses, the surface areas were fixed to these design values, and the flue gas temperature was varied to understand the effect of this temperature on the heat recovery and the system heating and cooling capacities. Thus, these analyses simulate the variation in performance of an already installed system with specific surface areas, as the flue gas temperature changes. Figure 3 shows the effect of the flue gas temperature on system capacities. It can be seen that even small changes in flue gas temperature have an appreciable effect on the various capacities. For example, as the flue gas temperature is increased from 246°F (slightly

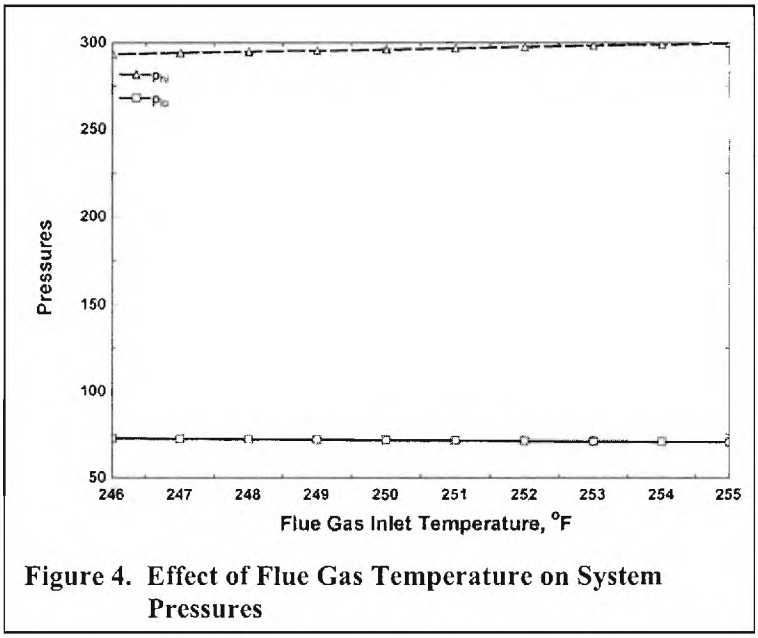

Figure 4. Effect of Flue Gas Temperature on System Pressures

below the nominal value) to 255°F, the heat recovered from the flue gas increases from 2.205 MW to 2.467 MW, while the cooling and heating capacities increase from 1.241 to 1.391 MW, and 3.481 to 3.894 MW, respectively. This can be explained on the basis of the system pressures, as shown in Figure 4. Thus, over the same range of flue gas temperature, the system low pressure decreases from 72.73 to 70.58 psia, while the high pressure increases from 293.4 to 299.6 psia. Thus, as expected, the pressure ratio over which the system operates increases as the flue gas temperature is increased, keeping other parameters constant. This increase in pressure ratio increases the "heat pumping" capability of the system. Figure 5 shows the variation in the concentrated and dilute solution concentrations over this same range of flue gas temperatures. Both concentrations decrease as the flue gas temperature increases. This is because a higher flue gas temperature results in a higher inlet and outlet temperature of the ammonia-water solution, which is achieved at lower concentrations (due to the higher water fraction – water has a higher boiling point than ammonia.) The difference in the concentrated and dilute solution concentrations also increases slightly as the flue gas temperature increases. For a constant pumped solution flow rate, this implies that more vapor is desorbed out of the solution at higher flue gas temperatures, leading to higher cooling and heating capacities. Also, the lower concentrations (and the higher solution temperatures) at the higher flue gas temperatures provide a larger driving temperature difference between the working fluid and the heated water at the absorber, leading to larger heating capacities.

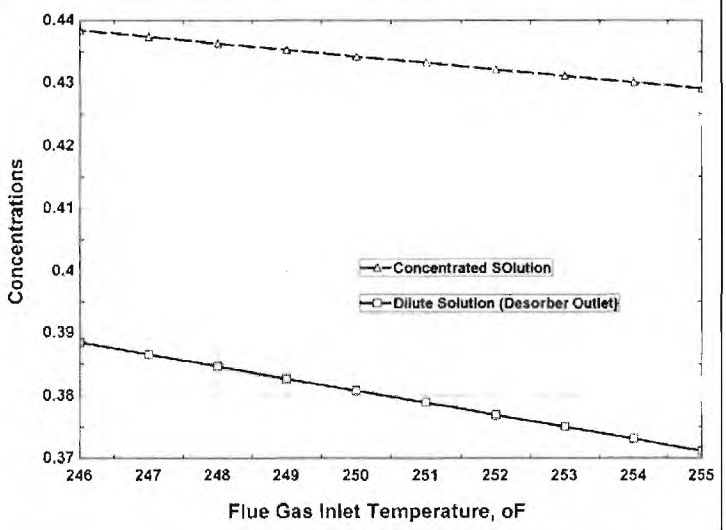

Figure 5. Effect of Flue Gas Temperature on System Concentrations

Depending on ALCOA's preferences between operating and first costs of such a system, a perhaps more interesting parametric analysis is the variation of the required surface area of the desorber with an increase in flue gas temperature, *keeping cooling capacity constant.* Thus, unlike the previous analysis, instead of using the higher flue gas temperature for increased

cooling capacity, this analysis investigates how the desorber size would decrease due to the increase in flue gas temperature. The cooling capacity is kept fixed at the nominal 1.275 MW for this analysis. Figure 6 shows the effect of flue gas temperature on desorber surface area. A mere 9°F increase in this temperature from 246°F to 255°F results in a decrease in the required surface area from 60,700 ft^2 to 27,600 ft^2 – a several-fold decrease. This finding highlights the importance of waste heat source temperature clearly.

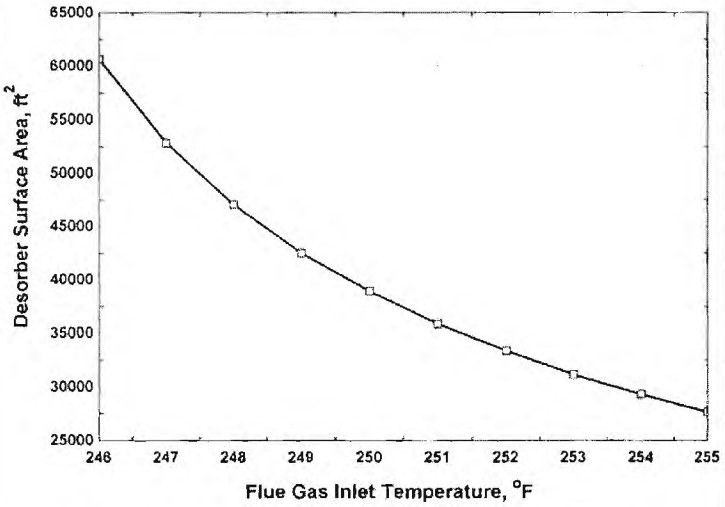

Figure 6. Effect of Flue Gas Temperature on Desorber Surface Area at Constant Cooling Capacity

This influence of flue gas temperature can be used to guide potential plant design decisions. For example, certainly, the 15°C temperature drop anticipated in the gas treatment plant is a significant matter that could essentially determine the viability of this heat recovery system. If the flue gas temperature decreases from the current assumed value of 248°F to 221°F due to additional gas treatment, it is unlikely that an absorption system, or for that matter, any thermally activated system can be implemented to meaningfully recover any heat of practical use. Therefore, the absorption system should be installed upstream of this gas treatment operation. On the other hand, if it were at all possible to decrease the dilution of the 960°C initial waste heat stream from a 9 to 1 ratio even to an 8.5 to 1 ratio, it would yield significant benefits for the heat recovery system. Also, if the process plant can be modified such that this dilution of the 960°C stream were conducted downstream of the heat recovery system, the possibilities available for heat recovery would increase by orders of magnitude, including the applicability of advanced multiple-effect systems, and perhaps even power generation/cogeneration systems from that hot waste stream.

On other possibility deserves mention here. It was stated above that the stated 80 MW of waste heat could not realistically be recovered from the subject waste stream because cooling

that stream substantially would limit its utility as a heat source. There is, however, a more complex absorption heat pump system known as a half-effect cycle that can recover heat down to very low temperatures. Thus, the initial heat recovery from the 248°F stream would be accomplished at pressures and ammonia-water saturation temperatures similar to those in the current single-effect cycle. When the waste heat stream temperature decreases beyond its usefulness to a single-effect cycle, a second desorber at a lower pressure would be used to recover heat at lower temperatures. This is the essence of the half-effect cycle that would increase the achievable cooling and heating capacities, although its implementation requires some other details that can be pursued if interest is expressed by ALCOA.

Effect of Hydronic Fluid Temperatures and Cooling-Only or Heating-Only Systems

The above analyses were conducted under the assumption that both chilled water and heated water are of interest as products from the heat recovery system. This requires that these hydronic fluid streams have relatively lower flow rates, so that an appreciable temperature change (55 to 45°F for the chilled water, 110 to 130°F for the heated water) can be achieved in both hydronic fluid streams. But the low flow rates and the large ΔTs required imply that a significant portion of the overall ΔT available between the heat source and the sink is being consumed by these coupling fluid ΔTs. If in certain instances, only cooling is of interest, then, the heating fluid can be run at a significantly higher flow rate, and perhaps with a ΔT of only 10°F or lower instead of the current 20°F requirement. This would reduce the required high pressure, or conversely increase cooling capacity for a given pressure ratio. From the same considerations, if or only heating is of interest, the cooling hydronic fluid flow rate could be increased, yielding higher overall system performance. A similar consideration applies to the hot hydronic fluid inlet of 110°F that was chosen in this study. This choice was made using the rationale that for any practical water heating application, heat could be extracted from the heated water only down to 110°F, below which it loses utility as a heating fluid. However, if generating hot water simultaneously is not a significant consideration, the hydronic fluid temperature could be decreased to even 100°F or close to the ambient temperature, to realize system performance improvements. These investigations can be conducted in future phases after the initial results from this study have been evaluated by ALCOA, and the preferred path to follow is identified.

CONCLUSIONS

A preliminary design and simulation of a single-effect ammonia-water heat pump was conducted as a heat recovery device for an aluminum smelter. The waste heat gas stream was assumed to be available at 248°F based on discussions with ALCOA. Ammonia-water was selected as the working fluid because of the potential intended applications over a wide range of ambient conditions and system capacities. It was found that the source temperature is the most significant factor in determining the ability to recover heat usefully. At the nominal conditions, based on several assumptions about system layout and coupling fluids, it was found that 2.26 MW of heat can be recovered from the waste heat stream to run the absorption system. This yields a chilled ethylene glycol-water stream at 44.5°F with a cooling capacity of 1.275 MW. Simultaneously, a second ethylene glycol-water stream can be heated from 110°F to 130°F for a heating capacity of 3.573 MW. Based on reasonable assumptions about the cost of electricity to generate this cooling without this waste heat recovery system, and the cost of natural gas to generate this heating with this heat recovery system, it was estimated that a savings of $186/hr of operation can be realized. No credit for avoidance of peak demand charges was taken in this estimate. When extrapolated to annual operation with a 75% capacity factor, it was found that annual savings of up to $1.2 million can be achieved in each implementation of such a heat recovery system. Caution must be exercised in evaluating the predictions of this "zeroth order" analysis, *because all these predictions were based on the one single input information from ALCOA regarding the flue gas temperature.* Local and plant-specific aspects would play a big role in whether such savings can be realized in an actual system.

The system described above is predicated on the installation of rather large heat exchangers that enable heat exchange to occur over very small temperature differences, whose sizes were estimated based on assumptions about component overall heat transfer coefficients. Again, these are very approximate estimates that would need to be refined considerably in future phases of this work, if ALCOA finds these preliminary results to hold potential for further investigation. The largest component in the system was, not surprisingly, found to be the waste gas to ammonia-water system coupling heat exchanger (the desorber). Although this will be a very large heat exchanger, minor increases in heat source temperature result in substantial reductions in the required heat exchanger size. Some potential strategies to enhance heat source temperature are pointed out in this study, along with the anticipated outcomes of such

temperature increases. Also, systems specifically tailored to chilling-only or water heating-only applications may hold potential for improved system performance. Georgia Tech would be glad to conduct more detailed analyses, system development, prototype fabrication and testing efforts for such heat recovery systems in future phases if desired by ALCOA.